

Article

Dynamic Characteristic Analysis and Experimental Study of a Direct-Acting Centrifugal Speed Limiter Based on Adams–Ansys

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Abstract

For the newly designed direct-acting centrifugal speed limiter, this paper employs a joint simulation method using ADAMS 2019 and ANSYS 2024, and combines the experimental data to analyze the dynamic characteristics and structural strength reliability of the speed limiter. The results show that the error between the simulated trigger angular velocity and the set value is only 1.31%, and the error with the mathematical model is 3.23%. The maximum structural stress is only 2.62% of the material yield strength, and the first six natural frequencies ranging from 201 to 377 Hz are much higher than the elevator operating frequency band, eliminating the risk of resonance. The relative errors between the experimental test data, theoretical models, and simulation results are all less than 6%, verifying the accuracy and reliability of the design. The speed limiter exhibits sensitive triggering and excellent dynamic stability, providing a methodological reference for the design and verification of elevator safety components.

Keywords: speed limiter; structural design; ADAMS; ANSYS; modal analysis

1. Introduction

In recent years, with the rapid urbanization and rising population density in China, elevators—essential vertical transportation equipment in modern buildings—have witnessed explosive demand growth [1,2]. As a core safety component in elevator safety protection systems, the speed limiter monitors car speed in real time and triggers safety braking [3]. It primarily operates on two principles: pendulum and centrifugal. The pendulum-type relies on the deflection of a gravity swing rod to sense speed changes, while the centrifugal-type uses radial displacement of rotating centrifugal weights for speed detection [4]. Over 60% of elevator accidents are directly linked to overspeed and loss of control, underscoring the criticality of reliable elevator speed monitoring and braking systems [5].

Scholars have achieved new progress in the structural innovation, dynamic characteristic analysis, and reliability optimization of centrifugal speed limiters. Lonkwc and Szydło [6] performed a performance comparison of counterweight and spring tensioning systems for traditional centrifugal speed limiters, and proposed an innovative spring-based rope tension adjustment method. Experimental verification showed that without



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the counterweight, the spring could effectively maintain stable rope tension—only a 3.5% decrease in spring tension was observed in a 356-day test, confirming the feasibility of this alternative solution. Gradziel MJ [7] studied a compact lowering device for rovers, which uses a centrifugal friction brake to release tethers at a stable speed for safe payload lowering. Tian et al. [8] proposed an integrated solution of a speed regulator and safety clamp: a friction wheel was integrated on the car top to simplify the synchronous belt drive structure; electromagnetic braking combined with a ratchet reduced the braking response time to 0.15 s, enhancing safety. Centtagoya et al. [9] developed a new biased mass speed limiter to address the wear and noise issues of traditional cam–follower speed limiters. Structural optimization reduced the number of moving parts from 12 to six, lowering the maintenance demands and costs. Multi-body dynamics simulation verified that the braking speed error was only $\pm 1.5\%$, and the braking speed deviation under a 0.5 mm joint gap was 40% lower than that of traditional designs—improving dynamic stability and providing an effective solution to traditional speed limiter drawbacks. Xie et al. [10] successfully developed a miniature centrifugal friction speed limiter (weight: 225 g) with a compact, lightweight structure. It adopted a symmetrical layout of three brake shoes and used a micro-arc oxidation aluminum drum to enhance performance and durability. Adams simulation analysis showed that under a variable driving torque of 0.017–0.15 N·m, the steady-state speed fluctuation rate was only 33.4%, verifying the model’s good adaptability to variable load conditions. The study also noted that reasonable symmetrical design could completely eliminate the interference of brake shoe gravity on performance, offering new ideas and methods for the precision control of micro speed limiters. Kutsepolenko et al. [11] performed a geometric analysis of a dual-rocker hinge mechanism in bidirectional centrifugal elevator speed limiters. The displacement–rotation relationships of the left and right rockers were derived, revealing inherent kinematic asynchronism between the two sides. Under nominal dimensions, a rotational angle deviation of approximately 4.89×10^{-3} rad occurs at the contact instant, causing asynchronous engagement of the bilateral components. This may degrade the bidirectional limiting performance into unidirectional operation. Accordingly, optimizing linkage dimension matching was recommended to suppress asynchronous motion and improve operational reliability. Zhang et al. [12] proposed and statically analyzed swing and guided sliding mechanisms for human–computer interaction. The nonlinear relationships among the external force, structural displacement and spring elongation were established, and their force–displacement characteristics were compared. The swing structure is simple but suffers from rapid force growth after activation, whereas the guided sliding structure effectively reduces force increment but introduces notch-induced peak force with higher structural complexity. Numerical validations demonstrate that the proposed design is applicable to passive force-limiting device development. In aerospace applications, Shahjahan et al. [13] systematically classified and compared typical torque limiters for electromechanical actuators (EMAs) in terms of working principles, performance and commercialization status. Friction-type and permanent-magnet torque limiters were identified as optimal candidates owing to their high torque density, superior high-speed performance and good reliability. Integrating torque limiters with driving motors was further discussed to reduce system volume and weight. For dynamic power regulation, electronic speed limiters exhibit prominent advantages. Targeting abrupt propeller load fluctuations of watercraft engines, Micheau et al. [14] developed a speed limiter based on feedback linearization and online Kalman load torque estimation, which regulates engine torque via precise air–fuel ratio adjustment. Experimental results show that the proposed controller maintains stable speed under partial load, sudden unloading and reloading conditions. Compared with conventional ECU fuel cut strategies, it effectively suppresses speed and noise fluctuations, improving system safety and comfort.

Centrifugal speed limiters are widely used due to their compact structure and sensitive response. However, the ratchet–pawl structure has a critical defect: when the release speed is reached, the pawl allows the ratchet to rotate one extra turn before triggering the brake. This causes the elevator to slide down by 1–3 m, posing a significant threat to safety. To address this issue, it is essential to design a speed limiter with high braking accuracy and sensitivity. Therefore, this paper proposes a direct-acting centrifugal speed limiter (DCSL) that meets such requirements. Adams–Ansys co-simulation analysis was conducted on the DCSL to verify the effectiveness of its design. Meanwhile, MATLAB 2023 was used for data processing to validate the accuracy and reliability of the DCSL’s mathematical model and simulation results. The research process is shown in Figure 1.

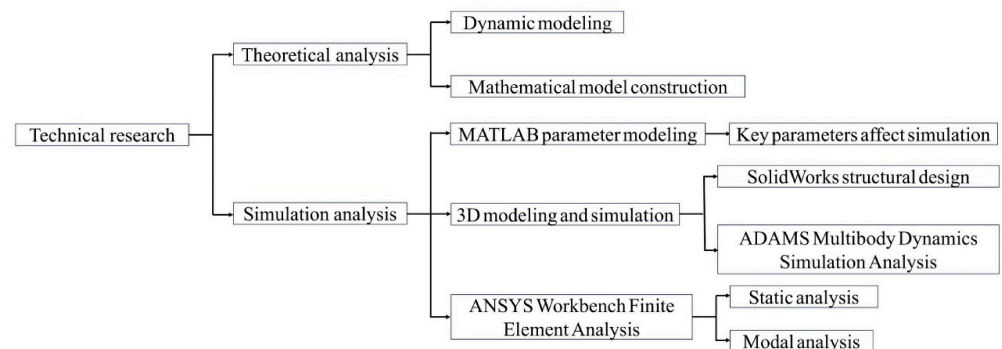


Figure 1. Research flow chart.

2. Structural Design

2.1. Establishment of 3D Model

Elevator speed limiters are mainly divided into pendulum-type and centrifugal-type according to the operating principles. The pendulum-type limiter works based on the resonance amplification effect, while the centrifugal-type limiter relies on the centrifugal force generated by movable masses. When such force overcomes the preload of the constraint spring, the masses swing outward and activate the braking mechanism [15].

Pendulum-type speed limiters have inherent deficiencies in scenario adaptability and dynamic reliability. Restricted by the resonance-triggered mechanism, they are only applicable to low-speed elevators with a rated speed below 1 m/s and cannot satisfy the overspeed protection requirements of high-speed elevators operating at 2.0–6.0 m/s [4]. Their discontinuous triggering characteristic easily causes speed overshoot. Moreover, such limiters are highly sensitive to machining and assembly errors, leading to inaccurate and unstable calibration of the triggering speed [16,17]. In addition, the absence of a reliable rope-clamping mechanism results in low braking efficiency and potential failure risks. The complex dynamic coupling of structural components also increases the difficulty of equipment maintenance and fault detection [18,19].

The composition structure of the direct-acting centrifugal speed limiter is shown in Figure 2, with a compact overall layout and clear functional zoning. The speed limiter is mainly composed of a box body, an end cap, a connecting rod, a trigger shaft, a trigger plate, a centrifugal block component, and a speed-regulating spring component. The force balance between the centrifugal block and the speed-regulating spring is used to monitor the elevator speed. During normal operation, the pre-tension force of the spring causes the centrifugal block to retract internally; when the elevator stalls, centrifugal force overcomes the spring force and drives the centrifugal block to swing outward. The mechanical linkage device pushes the trigger shaft to produce axial displacement, which is detected by sensors and converted into electrical signals. The hydraulic braking system is quickly activated

to achieve safe braking. This design abandons the intermediate transmission link and responds quickly.

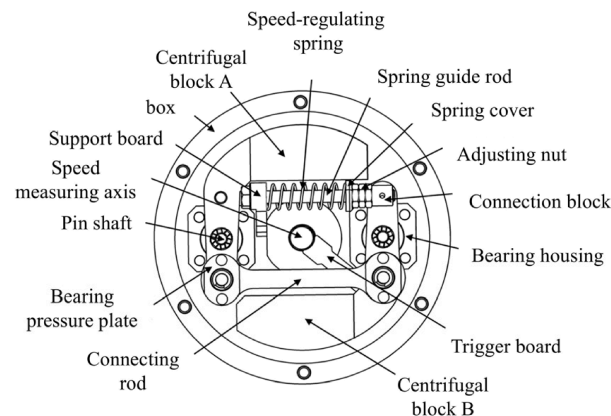


Figure 2. Structural schematic diagram of the direct-acting centrifugal speed limiter.

The working principle of traditional speed limiters is similar to the above, but the difference is that traditional speed limiters use mechanical linkage to clamp the ratchet claws into the rotating ratchet teeth slots. However, the ratchet claws have obvious defects. When the speed limiter reaches the critical speed, the claws need to rotate at least one revolution before triggering the brake. When the elevator stalls, the elevator car will continue to slide a distance of 1–3 m after rotating one revolution. The direct-acting centrifugal speed limiter used in this study will brake directly when the elevator stalls, reducing the buffering time in between and lowering the probability of danger.

2.2. Centrifugal Block Dynamics Analysis

To facilitate the analysis of the motion process of the direct-acting centrifugal speed limiter, the following assumption is made during the establishment of its dynamic model: only the centrifugal force generated by the centrifugal blocks is considered throughout the motion process, while the centrifugal effects of other components are neglected. Based on this assumption, centrifugal blocks A and B are treated as separate bodies, and coordinate systems xO_1y and xO_2y are established accordingly. The mechanical model of the direct-acting centrifugal speed limiter is shown in Figure 3.

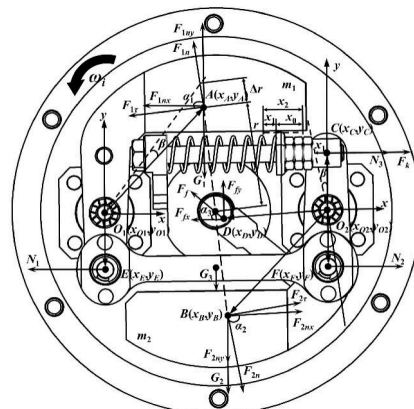


Figure 3. Mechanical model of direct-acting centrifugal speed limiter.

Due to the smooth movement of the speed limiter, the impact of slight vibrations on the system's motion is ignored; the influence of factors such as air resistance and friction during the movement of the speed limiter on the system's motion are neglected; the mass of the connecting rod, $G_3 = 0$, is neglected; when the motion of centrifugal block A reaches an

equilibrium state, according to the D'Alembert principle, there is a torque balance equation, $\sum M_{O1} = 0$. Due to the gravity balance of the speed limiter, the masses of the two centrifugal blocks take the same value, that is, $m_1 = m_2$ [5]. m_1, m_2 is the mass of centrifugal blocks A and B. According to the balance principle, the following can be obtained

$$(y_A - y_{O1})(m_1 + m_2)\omega_i^2(r + \Delta r) \cos \alpha_1 + (x_A - x_{O1})(m_1 + m_2)\omega_i^2(r + \Delta r) \sin \alpha_1 \\ = (y_C - y_{O2})[k(x_0 + x_1 + L_1 - L_2)] + (y_{O2} - y_D)\mu F_P \cos \alpha_3 + (x_{O2} - x_D)\mu F_P \sin \alpha_3 \quad (1)$$

Among them, ω_i is the angular velocity of the speed limiter, r is the distance from the axis to the center of mass of the centrifugal block when the speed limiter is not triggered, (x_{O1}, y_{O1}) is the O_1 coordinate, (x_A, y_A) are the centroid coordinates of centrifugal block A, (x_{O2}, y_{O2}) are the O_2 coordinates, (x_D, y_D) are the coordinates of point D, k is the stiffness of the speed-regulating spring, and L_1, L_2 is the free length and installation space length of the speed-regulating spring.

2.3. Dynamic Analysis of Trigger Plate

In practical applications, the speed limiter triggers an NPN normally closed proximity switch (hydraulic sensor) coaxially mounted on the outer shaft end, with a pre-tensioned spring positioned in front of the sensor. A compression spring is installed at the inner end, which requires compression during installation to secure the trigger plate. Under normal operation, the pre-tensioned spring exerts force on the trigger shaft via the push rod, a force greater than that of the compression spring, thereby securing the trigger card to the trigger shaft and preventing axial movement. At this stage, the centrifugal force of the centrifugal block is less than the tension of the speed-regulating spring, resulting in no radial displacement. When the centrifugal force of the centrifugal block exceeds the tension of the speed-regulating spring, it swings outward from the axis, causing the trigger plate to disengage from the groove. The trigger shaft then moves toward the speed limiter under the action of the pre-tensioned spring, further compressing the compression spring. Upon receiving a signal from the hydraulic sensor, the hydraulic valve is triggered to activate the braking system. The force distribution during trigger plate action is shown in Figure 4.

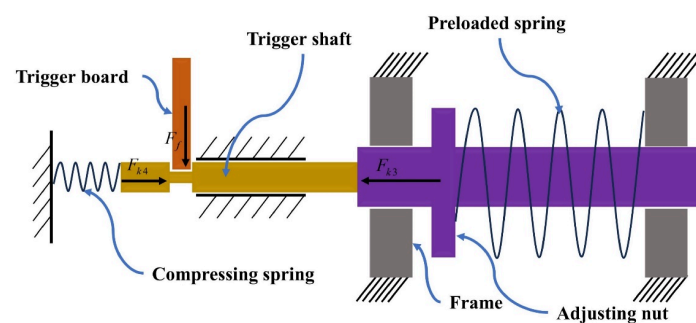


Figure 4. Force analysis diagram of trigger plate action.

When the speed limiter is triggered, the positive pressure between the trigger plate and the trigger shaft equals the difference between the pre-tightening force of the sensor pre-tightening spring at the outer end of the trigger shaft and the compression force of the trigger shaft compression spring at the inner end. From this, the mathematical equations for the spring precompression amount, speed limiter angular velocity, spring stiffness, and centrifugal block mass can be derived, as shown in equation [5].

$$x_0 = \frac{1}{k(y_C - y_{O2})} \{ (m_1 + m_2)\omega_i^2(r + \Delta r)[(y_A - y_{O1}) \cos \alpha_1 + (x_A - x_{O1}) \sin \alpha_1] - \\ \mu(k_3 x_3 - k_4 x_4)[(y_{O2} - y_D) \cos \alpha_3 + (x_{O2} - x_D) \sin \alpha_3] \} - (x_1 + L_1 - L_2) \quad (2)$$

3. Simulation Analysis

3.1. Adams Dynamic Simulation Analysis

Multi-rigid-body system dynamics is based on the rigid-body assumption, focusing primarily on the overall kinematics and dynamic performance of the system. Its neglect of component elastic deformation reduces model complexity, making it suitable for dynamic characteristic analysis of high-speed and high-inertia mechanical systems. In the elevator speed limiter design process, the multi-rigid-body model can accurately describe the contact separation process between the centrifugal block and the trigger plate, and clarify the influence law of key parameters on the braking speed [9]. To enable more efficient ADAMS multi-body dynamics simulation, the 3D model of the direct-acting centrifugal speed limiter was first reasonably simplified.

By simplifying, the model shown in Figure 5 is obtained, which retains key structures and functions, eliminates redundant details, reduces complexity, and improves simulation efficiency and accuracy. In ADAMS simulation, the axis of the speed limiter is set as the origin and the negative X-axis direction is the direction of gravity. The quality attributes and constraints of the components are defined, and the important constraints are shown in Table 1. The constraint position is shown in Figure 5.

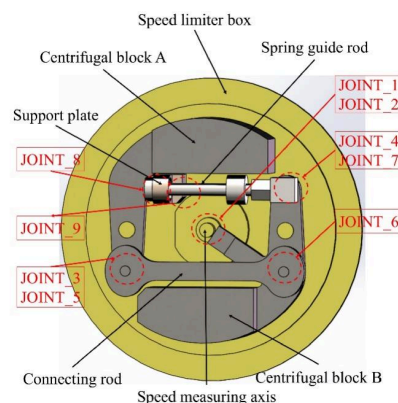


Figure 5. Simplified model of direct-acting centrifugal speed limiter.

Table 1. Constraints between important components.

Sports Vice Name	Constraint Type	Component 1	Component 2
JOINT_1	Fixed vice	Speed measuring axis	Ground
JOINT_2	Revolute joint	Speed limiter box	Speed measuring axis
JOINT_3	Revolute joint	Centrifugal block A	Speed limiter box
JOINT_4	Revolute joint	Centrifugal block B	Speed limiter box
JOINT_5	Revolute joint	Connecting rod	Centrifugal block A
JOINT_6	Revolute joint	Connecting rod	Centrifugal block B
JOINT_7	Revolute joint	Spring guide rod	Centrifugal block B
JOINT_8	Revolute joint	Support plate	Speed limiter box
JOINT_9	Mobile deputy	Spring guide rod	Support plate

Based on the theory of rigid-body system dynamics, a parameterized virtual prototype is established using ADAMS 2019 software to verify the mathematical model of the triggering conditions obtained from dynamic analysis

$$F_{centrifugal} = mr\omega_{release}^2 = F_{spring,preload} + kx \quad (3)$$

This model describes the state where the centrifugal force and spring preload of the speed limiter reach equilibrium at the critical speed. Among them, represents the mass of

the centrifugal block, represents the rotation radius, represents the spring stiffness, and represents the spring compression amount.

In the simulation analysis phase, the drive part employs the STEP function for speed definition. Per the mathematical model's operating conditions, the speed limiter's release angular velocity is 34.29 rad/s. Thus, the drive function is set as STEP (time, 0,0,10,−34.29), defining a linear speed variation from 0 to 10 s, during which the speed increases linearly from 0 to −34.29 rad/s. The drive function's angular velocity variation curve is shown in Figure 6.

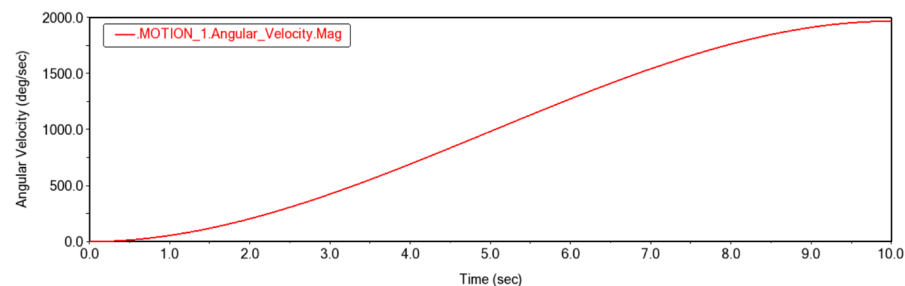


Figure 6. Drive function curve.

This simulation sets the start time to 0 and end time to 10 s. The speed increases linearly from 0 rad/s at $t = 0$ s to −34.29 rad/s at $t = 10$ s; the negative speed value indicates counterclockwise rotation of the component. This rotation direction matches that in the speed limiter's dynamic theoretical model, guaranteeing the accuracy and reliability of the simulation.

3.2. Simulation Result Analysis

Per the dynamic characteristics requirements of the direct-acting centrifugal speed limiter, ADAMS is configured with a simulation termination time of 10 s and 100,000 steps (step size: 0.1 ms) to accurately capture the high-speed triggering transient response. After simulation completion, the ADAMS/Post Processor module is accessed to extract the dynamic parameters. Figures 7–9 respectively illustrate the time-dependent deformation, force, and angular velocity of the speed-regulating spring.

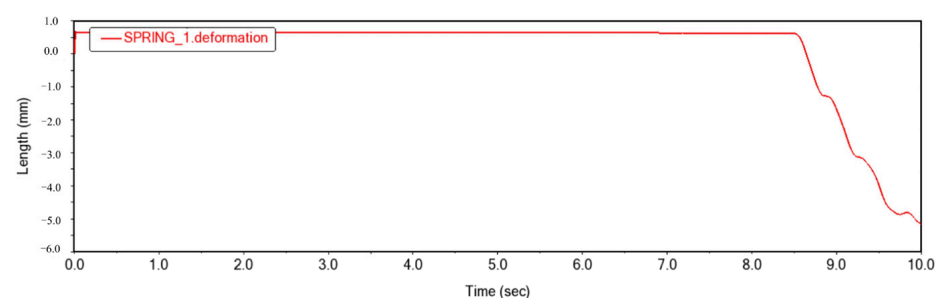


Figure 7. Curve of deformation of speed-regulating spring over time.

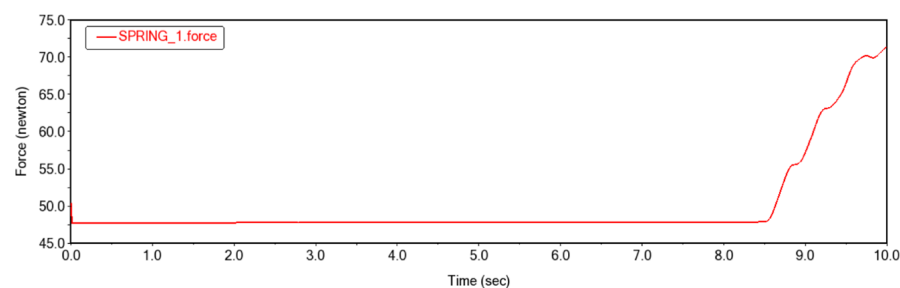


Figure 8. Time-varying curve of force on the speed-regulating spring.

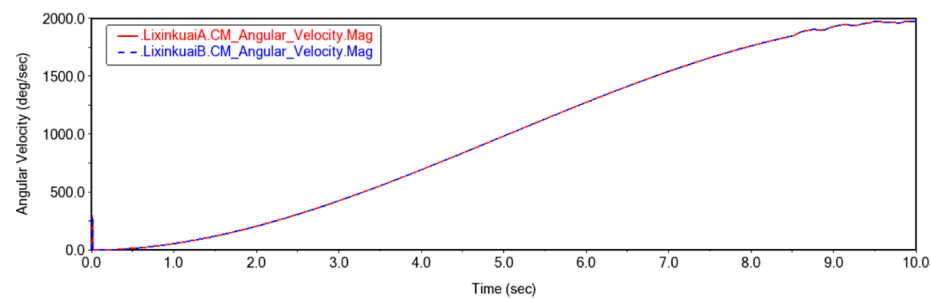


Figure 9. Time-varying curve of angular velocity of two centrifugal blocks.

From Figure 7, in the early simulation stage, the compression amount of the speed-regulating spring remains basically at 0 mm, indicating that in the untriggered state of the speed limiter, the spring is in a normal precompressed state without external deformation. When approaching 8.5 s, the compression starts to increase nonlinearly, reaching 2.2 mm at 9.08 s. At this point, the speed limiter successfully releases the trigger shaft, the spring compression reaches the preset value, and the braking mechanism is triggered accordingly.

From Figure 8, in the initial simulation stage, the force on the speed-regulating spring stabilizes at approximately 50 N, consistent with the spring compression trend, confirming stable spring force in the speed limiter's untriggered state. As the time approaches 8.5 s, the spring force starts to increase rapidly, reaching roughly 70 N at 10 s. This change aligns with the spring compression variation law, verifying the speed limiter's triggered dynamic response. Meanwhile, it indicates the spring plays a critical role in the braking process: its force increase enables the speed limiter's braking control.

From Figure 9, the angular velocity of the two centrifugal blocks increases linearly with time, highly consistent with the preset driving function. During the 0–9.08 s period, their angular velocity increases uniformly from 0 rad/s to 33.84 rad/s (corresponding to 1938.78/s). At the triggering time $t = 9.08$ s, the corresponding relative error of the angular velocity is calculated as follows:

$$\delta = \frac{|33.84 - 34.29|}{34.29} \times 100\% = 1.31\% \quad (4)$$

Analysis on the angular velocity variation in two centrifugal masses A and B with time shows that from 0 to 9.08 s, the angular velocity rises uniformly from 0 rad/s to the braking threshold of 33.84 rad/s, which matches the designed kinematic performance in the acceleration stage. At the triggering instant $t = 9.08$ s, the deviation between the measured angular velocity and the preset value 34.29 rad/s of the driving function is merely 1.31%, while its error relative to the braking speed from the mathematical model is 3.23%.

The discrepancies between the simulation and theoretical data of the speed limiter mainly stem from model simplification and idealized simulation configurations. Neglecting partial centrifugal effects of components and simplifying linkages in the dynamic model underestimates the system moment of inertia, yielding lower simulated trigger angular velocity. Meanwhile, frictionless and clearance-free ideal joints adopted in ADAMS deviate from the real operating conditions, delaying trigger response and causing a 1.31% simulation error. In addition, both simulation and theoretical derivation omit air resistance and structural damping, leading to a 3.23% deviation from the mathematical model. Such errors are far below the 10% upper limit permitted by the design criteria and the tolerance specified in GB/T 7588-2020 [20], which verifies the accuracy of the mathematical model and the rationality of simulation settings.

3.3. Ansys Modal Analysis

3.3.1. Model Simplification and Material Definition

Prior to modal analysis and static analysis in ANSYS Workbench, geometric simplification and material parameter definition shall be performed on the 3D model of the direct-acting centrifugal speed limiter to ensure the high accuracy and computational efficiency of the simulation model. Based on the constructed simplified model (Figure 4), the speed-regulating spring assembly is integrated into the geometric structure; the geometric simplification adheres to the principle of “retaining key force transmission paths and removing non-load-bearing details”, only preserving core load-bearing components including the box body, centrifugal blocks, trigger plate, connecting rods, and speed-regulating spring.

The speed limiter adopts impact-resistant 304 stainless steel, featuring high strength and a high safety factor that enables it to meet the basic operational requirements of the speed limiter. The material model parameters are shown in Table 2.

Table 2. Parameters of 304 stainless steel model.

Modulus of Elasticity (GPa)	Poisson's Ratio	Density (kg/mm ³)	Compressive Yield Strength (MPa)
193	0.31	7.75×10^{-6}	207

3.3.2. Grid Division

Mesh division is a critical step in finite element analysis, directly influencing the balance between computational accuracy and efficiency. As shown in Figure 10, the model adopts a 3 mm mesh size. Preliminary studies indicate that modal analysis fails to converge for 1 mm or 2 mm mesh sizes due to excessive computational complexity, while 4 mm or 5 mm meshes result in relatively few elements. The total deformation results for the five mesh sizes are compared, as shown in Table 3. Ultimately, a 3 mm mesh balances computational accuracy and solution.

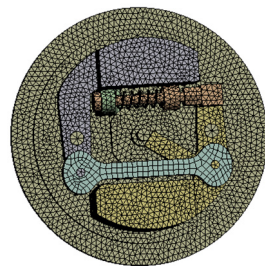


Figure 10. Grid division model of direct-acting centrifugal speed limiter.

Table 3. Calculation results corresponding to different grid sizes.

Grid Size	Number of Grids	Number of Nodes	Result/mm
1 mm	1,037,727	1,766,244	7.9950×10^{-3} mm
2 mm	203,474	350,509	7.9966×10^{-3} mm
3 mm	72,372	128,830	7.9972×10^{-3} mm
4 mm	33,649	64,659	7.7931×10^{-3} mm
5 mm	20,001	39,973	7.6816×10^{-3} mm

Structurally, the speed limiter housing and centrifugal blocks exhibit strong geometric symmetry; a uniform mesh prevents misjudgment of local stress concentrations. The contact surface between the trigger plate and speed-regulating spring has gentle curvature, where a 3 mm mesh suffices to capture edge stress gradients—while reducing element count and

enhancing efficiency. Based on the direct-acting centrifugal speed limiter's structural characteristics, a global uniform mesh size strategy is adopted: a 3 mm tetrahedral-dominant mesh is applied to all components via ANSYS Workbench's Mesh module, as shown in Figure 10.

3.3.3. Boundary Conditions and Load Settings

The strength and stiffness of the structure under the triggering critical state are mainly verified through statics, and its mathematical model is based on the basic control equations of elasticity. By solving the overall equilibrium equation:

$$F_1 = [K]u \quad (5)$$

Among them, $[K]$ represents the overall stiffness matrix of the speed limiter, u represents the node displacement vector, and F_1 represents the centrifugal load and gravity. The load conditions are set based on the triggering critical state, including centrifugal loads generated by angular velocities ranging from 0 to -34.29 rad/s, as well as gravity along the X-axis; applying a fixed support at the speed-measuring wheel axle end, i.e., $UX = UY = UZ = 0$, allows the speed limiter housing to rotate only around that axis. By solving this equation, the stress, strain, and deformation distribution of the structure can be obtained to verify whether it meets the strength and stiffness design criteria.

3.4. Modal Results Analysis

Modal analysis is used to identify the inherent vibration characteristics of structures in order to avoid resonance risks within the operating frequency range. Perform modal analysis under static analysis to solve the problem of model eigenvalues:

$$([K] - \omega_i^2[M])\{\phi_i\} = 0 \quad (6)$$

Among them, $[M]$ is the mass matrix, and ω_i and $[\phi_i]$ are the natural circular frequencies and corresponding mode vectors, respectively. Focus on the first six natural frequencies to ensure that they are significantly higher than the possible external excitation frequency range of the elevator system (5–50 Hz), in order to ensure dynamic stability. The first six natural frequencies are shown in Table 4, and Figure 11 respectively represent the cloud maps of the first six vibration modes.

Table 4. First six natural frequencies of direct-acting centrifugal speed limiter.

Modal order	1	2	3	4	5	6
Natural frequency/Hz	201.82	232.17	255.68	302.69	373.03	376.61

According to Table 2 and Figure 11a–c, the first-order natural frequency is 201.82 Hz, corresponding to the bending vibration of centrifugal block A in the counterclockwise direction, indicating that centrifugal block A is the most easily excited component of the system under dynamic conditions. The second-order natural frequency increases to 232.17 Hz, presenting an antisymmetric vibration mode that reflects an instability pattern of the system under high-energy input. The third-order natural frequency is 255.68 Hz, featured by synchronous oscillation of centrifugal blocks A and B, which verifies the dynamic consistency of the symmetric design and shows that the structure maintains orderly coordinated vibration instead of disordered flutter in the high-frequency region far beyond the working frequency, and the mode shapes become more complicated with the rise in mode order. As shown in Figure 11d–f, the fourth-order natural frequency is 302.69 Hz, mainly corresponding to the coupled vibration of clockwise bending vibration of

centrifugal block B and axial expansion–contraction of the speed-regulating spring, marking the transformation of the vibration mode from the oscillation of large rigid components to compound vibration involving elastic elements (springs). The fifth and sixth modes, with close natural frequencies, are high-order local modes of the system, where the vibration energy is mainly confined to the speed-regulating spring subsystem, indicating that the spring dominates the dynamic response of the system at extremely high frequencies.

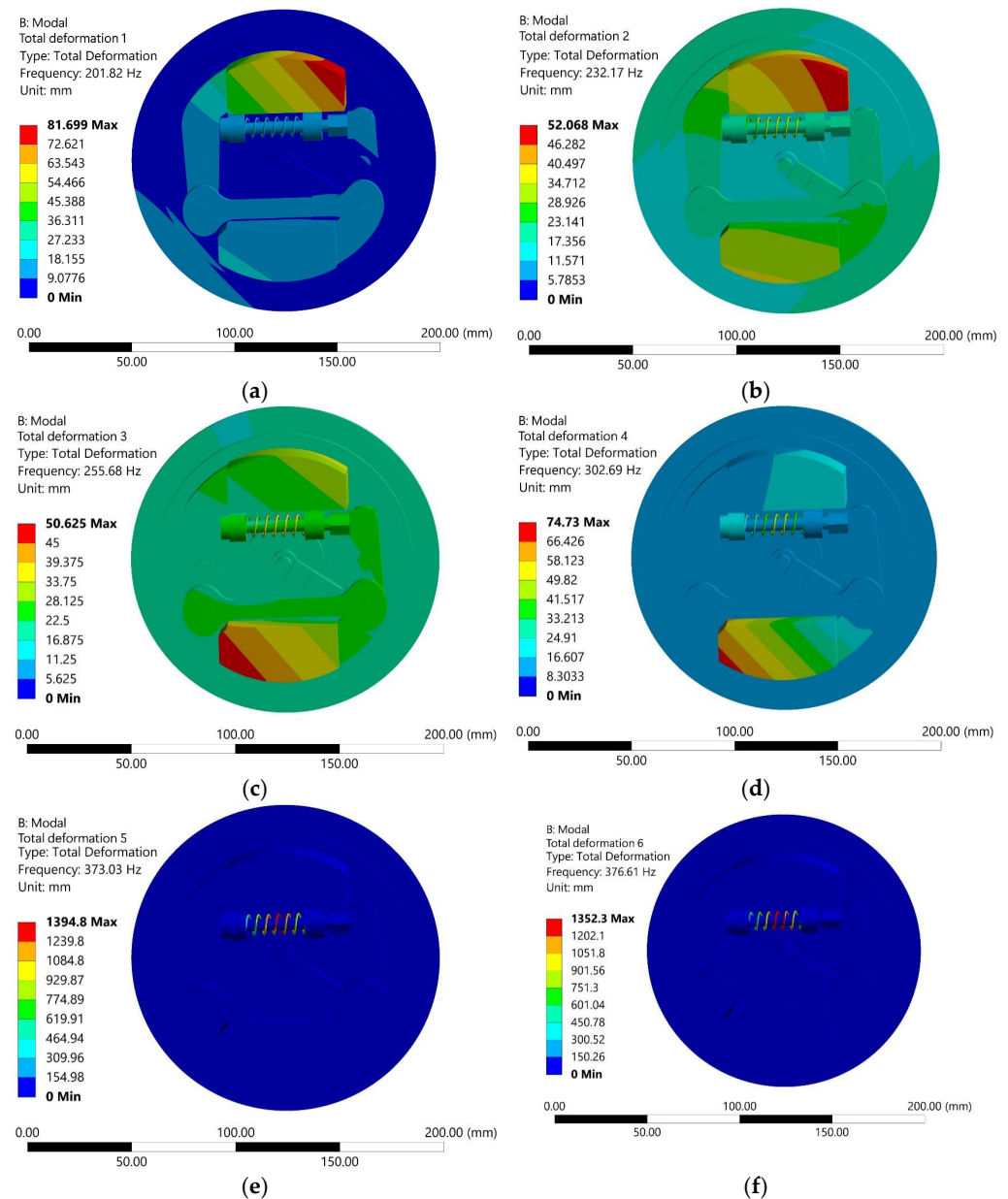


Figure 11. Diagram of the first three modes of vibration. (a) First-order mode shape diagram; (b) second-order mode shape diagram; (c) third-order mode shape diagram; (d) fourth-order mode shape diagram; (e) fifth-order mode shape diagram; (f) sixth-order mode shape diagram.

The normal operating frequency band of elevators ranges from 5 to 50 Hz [21]. The minimum natural frequency of this speed limiter reaches 201.82 Hz, far exceeding the 1.5-fold frequency design requirement specified in GB/T 7588-2020 and avoiding resonance risks. Elevator operation involves multiple excitation sources including electromagnetic engagement, braking impact and guide rail friction [22]. Modal analysis reveals that the first-order mode only brings negligible slight wear risk; the second- and third-order modes

maintain structural dynamic balance, while higher-order modes possess vibration isolation capacity. The overall structural dynamic stability meets the standards, and all local adverse effects are within allowable engineering ranges.

3.5. Static Analysis of Results

ANSYS Workbench-based static analysis, accounting for gravity, assesses the speed limiter's mechanical performance under actual operating conditions [23]. With applied loads and structural calculations, Figure 12 shows the overall deformation, maximum elastic strain, and maximum principal stress contour distribution of the speed limiter.

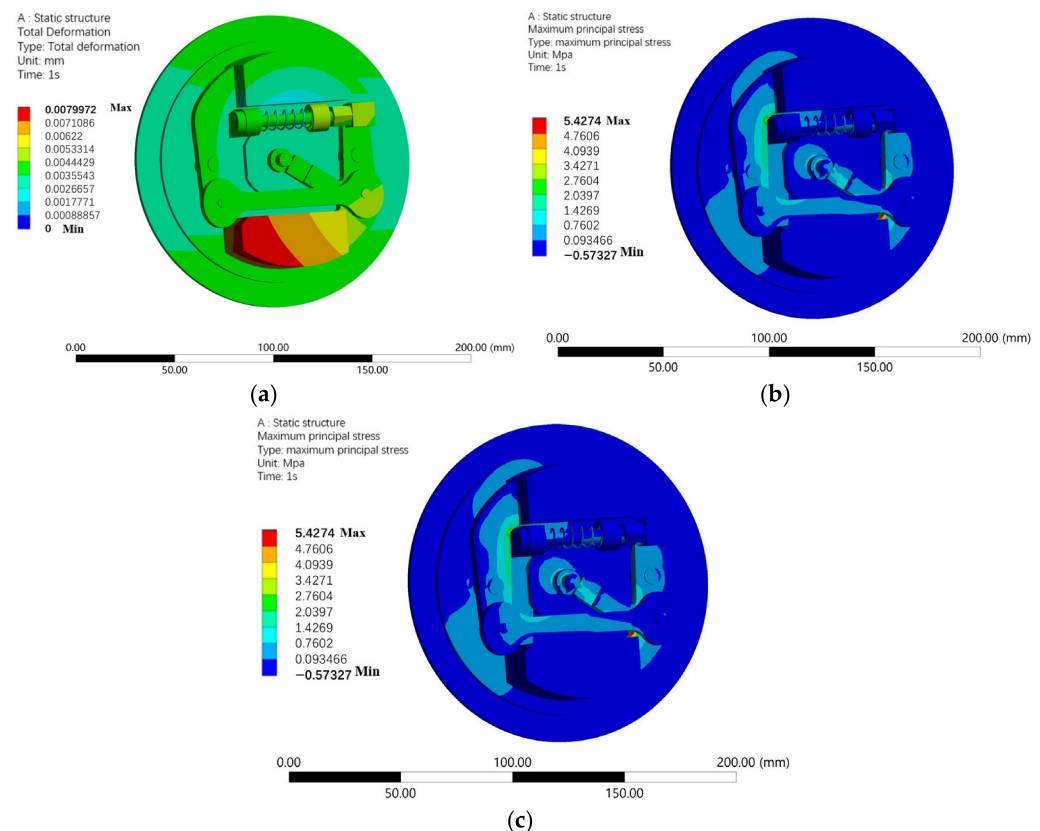


Figure 12. Static analysis results of direct-acting centrifugal speed limiter. (a) Deformation diagram; (b) strain diagram; (c) stress diagram.

From Figure 12a, the speed limiter's overall deformation concentrates on centrifugal block B's outer edge and the trigger plate, with a maximum of 7.9972×10^{-3} mm. Block B (core load-bearing part) bears maximum centrifugal force at high speed, causing slight radial displacement; the trigger plate deforms locally due to dynamic impact when detaching. This shows minimal deformation and good structural rigidity under design conditions.

From Figure 12b, the speed limiter's maximum principal elastic strain is 2.5874×10^{-5} mm/mm, concentrated at the weld between centrifugal block B and the trigger plate—caused by periodic bending moment from rotation. The overall elastic strain stays low, showing selected materials and structure control deformation well, ensuring reliable performance and long life, and verifying the rationality of its contact force transmission path.

From Figure 12c, the speed limiter's maximum principal stress is 5.4274 MPa, at the weld between centrifugal block B and the trigger plate. This is only 2.62% of 304 stainless steel's yield strength (207 MPa), showing extremely low stress under design conditions, high structural safety, and uniform stress distribution with no obvious concentration.

ANSYS analysis shows the speed limiter's total deformation, elastic strain and stress meet the design requirements. Its max deformation (7.9972×10^{-3} mm) reflects good stiffness, avoiding trigger delay/mis-operation and meeting elevator safety component standards. The strain concentrates at key nodes 10^{-5} order, the material stays elastic and the max stress is under 3% of the material yield strength, ensuring a high static load safety margin.

Modal analysis reveals dynamic stability and risk points, while static analysis verifies strength/stiffness compliance. Together, they confirm the speed limiter meets static load-bearing and dynamic anti-vibration requirements, satisfying elevator safety system needs.

4. Results and Discussion

4.1. Data Processing

A comprehensive test platform, as shown in Figure 13, was built to measure the critical release rotational speeds of the two limiters under spring precompression ranging from 0 to 10 mm, with the results listed in Table 5. The critical speeds of the direct-acting centrifugal speed limiter were compared with the theoretical and simulated values. Ten repeated tests were conducted for each precompression under simulated real elevator conditions: the test cabinet was rigidly fixed, and the driving end only rotated around its shaft. Triggering was judged by the signal mutation of proximity switches. The increase in the linear motor speed in repeated trials guaranteed reliable experimental data.

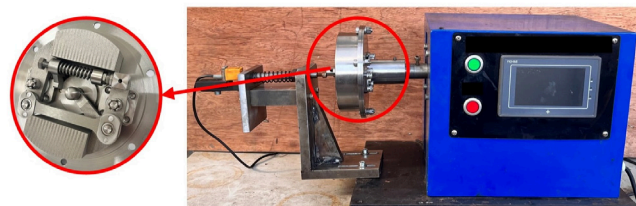


Figure 13. Data acquisition diagram of direct-acting centrifugal speed limiter.

Table 5. Release speed of direct-acting speed limiter.

Compression Amount/mm	Speed Limiter Release Speed/(r·min ⁻¹)									
	Group One	Group Two	Group Three	Group Four	Group Five	Group Six	Group Seven	Group Eight	Group Nine	Group Ten
0	310.25	314.85	318.42	315.91	311.77	316.33	312.52	317.69	313.50	319.14
2	331.52	336.84	332.76	338.18	334.90	330.64	337.22	335.48	333.96	339.08
3	342.56	347.84	341.92	345.76	348.32	340.64	346.12	343.28	349.04	344.60
4	361.56	368.24	363.78	366.92	360.84	367.50	362.18	365.36	369.40	364.62
6	391.48	396.72	393.26	398.54	390.86	397.18	394.62	395.90	392.34	399.02
8	411.56	418.24	413.78	416.92	410.84	417.50	412.18	415.36	419.40	419.62
10	431.72	438.46	433.98	436.54	430.82	437.16	432.58	435.34	439.20	434.68

Based on principal component multiple linear regression, orthogonal transformation extracts data features—preserving original info and solving rank defects from small samples. It better reflects multi-data coupling than univariate analysis predictions in aligning with reality [24], and the model analyzes the preload's impact weight on the release speed, supporting parameter optimization.

In MATLAB 2023, first use PCA to extract features from each experimental dataset, and then select principal components with a 93% cumulative contribution rate to build a regression model. The scatter plot of the experimental data is shown in Figure 14.

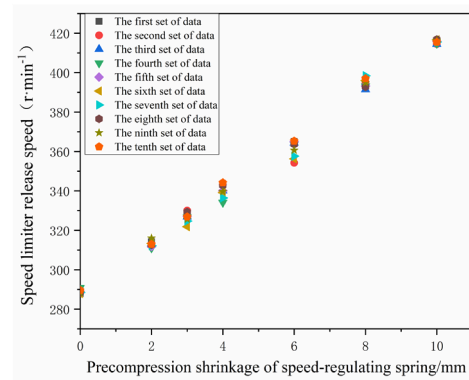


Figure 14. Scatter plot of experimental data.

From Figure 14, as the preloading amount changes continuously (0, 2, 3, 4, 6, 8, 10 mm), the speed limiter's release speed shows a significant linear upward trend; the scatter plot data points indicate a clear linear correlation between the two. Using a principal component multiple linear regression model, set $\alpha = 0.05$ to build a hypothesis test framework and get the fitting data [25]. Based on the mathematical model, obtain the theoretical, fitted and Adams release speeds under different preloading conditions, laying a theoretical foundation for subsequent simulation–experiment comparison.

According to Table 6, within a 0–10 mm speed-regulating spring precompression reduction, the max absolute error (10 mm) is 25.84 r/min, max relative error (0 mm) 5.97%, and min (2 mm) 0.31%. All relative errors are <10%, meeting the design requirements and GB/T7588-2020 [20] ($\leq 15\%$), proving high model reliability.

Table 6. Comparison of theoretical calculation values, experimental values, and Adams values.

Precompression Shrinkage/mm	Theoretical Release Speed/(r·min ^{−1})	Fit Release Speed/(r·min ^{−1})	Adams Releases the Rotational Speed/(r·min ^{−1})
0	273.98	293.35	283.02
2	315.33	316.35	314.38
3	334.09	328.15	330.87
4	351.86	341.68	346.71
6	384.93	363.05	375.45
8	415.38	396.83	406.77
10	443.75	418.91	429.87

The prediction accuracy of the model was quantified by calculating the relative error δ and absolute error Δ .

$$\delta = \frac{|\omega_{\text{experiment}} - \omega_{\text{theory}}|}{\omega_{\text{theory}}} \times 100\% \quad (7)$$

$$\Delta = |\omega_{\text{experiment}} - \omega_{\text{theory}}| \quad (8)$$

4.2. Verification of Mathematical Model

Regression analysis of the experimental data yields fitted data consistent with reality, verifying the direct-acting centrifugal speed limiter's engineering applicability. It analyzes the error distribution and causes of three release speeds (Table 6), and quantifies model accuracy via errors. The data are shown in Figure 15.

From Figure 15, the theoretical and experimental release rates both rise linearly with preloading—the trends align and curves match, although the theoretical curve's slope is slightly higher. The difference between the theoretical and experimental values is

0.31–5.97%, and the difference between the theoretical and Adams values is 0.29–3.29%. All relative errors are <10%, verifying model accuracy/reliability and supporting speed limiter structural parameter optimization.

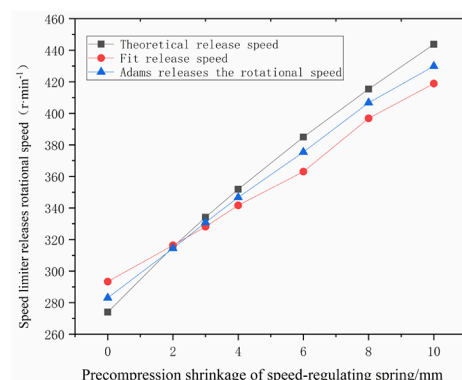


Figure 15. Data comparison curve.

Table 6 and Figure 15 reveal regular error variations among the three datasets with the spring precompression increasing from 0 mm to 10 mm. For a precompression of 0–4 mm, the theoretical, experimental fitting and ADAMS simulation values show excellent consistency. The relative error between the theoretical and experimental results is below 4.1%, while that between the theoretical and simulation results is within 3.0%. The low spring force in this range enables the centrifugal mass to trigger the limiter at a moderate speed. Minor deviations are attributed to idealized model assumptions, including neglected friction, rigid connections in theoretical calculation and undamped kinematic pairs in ADAMS simulation, which are well consistent with the actual test conditions.

In the 6–10 mm precompression range, data deviations increase obviously, and the measured triggering speed is persistently lower than the theoretical value. The absolute error rises from 21.88 r/min (6 mm) to 25.84 r/min (10 mm), and the relative error increases from 5.68% to 5.97%, still complying with error tolerance standards. In this range, the ADAMS simulations agree better with the theoretical calculations, with a relative error of merely 2.46–3.29%. Two core reasons cause the above discrepancies. First, larger precompression increases spring tension, requiring a higher speed and centrifugal force for limiter triggering. Under high-speed and high-load conditions, the hinge friction of centrifugal masses is magnified, and the friction torque rises with the spring force-dominated contact pressure. Ignoring friction in both the theoretical calculation and simulation leads to higher predicted triggering speeds than the experimental values. Second, a large spring deformation produces nonlinear behaviors including hysteresis and rate-dependent stiffness, which cannot be characterized by theoretical derivation and the linear ADAMS STEP driving function. In addition, prototype machining and assembly tolerances further amplify nonlinear errors under large precompression.

Across all precompression conditions, the ADAMS curves fall between the theoretical and experimental curves. Despite ideal constraint settings, ADAMS incorporates geometric nonlinearities such as inertial coupling and large linkage rotation, making it more practical than pure theoretical models, except for the exclusion of structural damping and friction. Model error grows with precompression, with optimal accuracy under small and medium preload conditions, and can be improved via experimental calibration and friction compensation for high-preload working conditions. All measured errors meet the design thresholds, verifying the validity of the proposed model and the operational reliability of the speed limiter within the tested preload range.

4.3. Comparison with Traditional Pawl Speed Limiter

To investigate the engineering applicability of the designed direct-acting centrifugal speed limiter (DCSL), a comparative analysis against conventional pawl-ratchet limiters is carried out. Traditional limiters rely on pawl-ratchet meshing for braking; at high speeds, the pawl often misses tooth grooves and engages after one tooth interval, raising the actual braking speed above the preset threshold and causing severe response delay. Eliminating the pawl-ratchet assembly and adopting direct transmission, the proposed DCSL enables centrifugal masses to push the trigger plate directly for braking, thoroughly removing the inherent delay and delivering superior triggering performance.

To further verify the superiority of the direct-acting centrifugal speed limiter, the release speed of the traditional pawl-type speed limiter was tested via the test platform described in Section 4.1. The test data are listed in Table 7.

Table 7. Release speed of pawl centrifugal speed limiter.

Compression Amount/mm	Speed Limiter Release Speed/(r·min ⁻¹)									
	Group One	Group Two	Group Three	Group Four	Group Five	Group Six	Group Seven	Group Eight	Group Nine	Group Ten
0	310.25	314.85	318.42	315.91	311.77	316.33	312.52	317.69	313.50	319.14
2	331.52	336.84	332.76	338.18	334.90	330.64	337.22	335.48	333.96	339.08
3	342.56	347.84	341.92	345.76	348.32	340.64	346.12	343.28	349.04	344.60
4	361.56	368.24	363.78	366.92	360.84	367.50	362.18	365.36	369.40	364.62
6	391.48	396.72	393.26	398.54	390.86	397.18	394.62	395.90	392.34	399.02
8	411.56	418.24	413.78	416.92	410.84	417.50	412.18	415.36	419.40	419.62
10	431.72	438.46	433.98	436.54	430.82	437.16	432.58	435.34	439.20	434.68

Figure 16 compares the averaged data from ten groups of tests with the varying spring precompression in Tables 5 and 7. The critical release speeds of both limiters rise linearly with precompression, demonstrating consistent design principles for centrifugal speed limiters. At identical precompression, the pawl-ratchet limiter triggers at a higher speed; the direct-acting limiter responds at a lower rotational speed and is more sensitive to spring precompression variation. Benefiting from shorter transmission paths and fewer friction interfaces, the direct-acting design exhibits a smaller trigger speed drift and better long-term stability, which confirms its structural reliability.

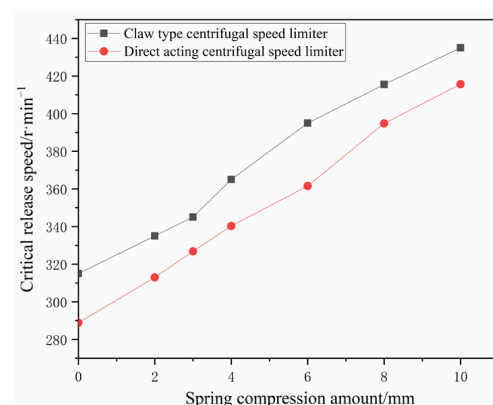


Figure 16. Comparison of data between direct-acting and pawl-type speed limiters.

From the passenger perspective, the instant triggering feature of the direct-acting centrifugal speed limiter eliminates the response delay risk of pawl-type devices. Once overspeed is detected, it initiates braking immediately, shortening the elevator car sliding

distance, cutting the peak emergency speed, reducing safety gear deceleration, and lowering passenger injury risks inside the car.

5. Conclusions

To address the large braking error and low sensitivity of traditional pawl-type centrifugal speed limiters, this paper proposes a direct-acting one (DCSL), conducting dynamic modeling, ADAMS simulation, ANSYS analysis and experiments. The results show its reliable mathematical model, ultra-low braking error, and improved elevator safety and reliability.

- (1) The mathematical model of the DCSL boasts high accuracy and excellent dynamic performance: it can accurately predict operational characteristics. Multi-body dynamics simulations via ADAMS show that the error between the simulated braking speed and the required operating speed is only 1.31%, and the error with the mathematical model is 3.23%, verifying the speed limiter's high braking accuracy and the model's reliability.
- (2) The structural stiffness, strength, and dynamic stability are reliable: the ANSYS static and modal analysis results show that the maximum stress at the welded joint between centrifugal block B and the trigger plate is 5.4274 MPa, accounting for only 2.62% of the material's yield strength, with a maximum deformation of merely 7.9972×10^{-3} mm. Modal analysis extracts the first six natural frequencies of the speed limiter, all significantly higher than the elevator's operating frequency range, effectively avoiding resonance risks. This verifies the rationality of the speed limiter's anti-vibration design under high-speed rotation conditions and ensures operational dynamic stability.
- (3) Wide engineering applicability: A test platform and hardware testing system were built, completing 10 groups of 70 repeated tests. Multiple linear regression analysis was conducted on the experimental data, verifying that the errors between the test values and mathematical model values under different precompression amounts of the speed-regulating spring range from 0.31% to 5.97%, and the braking speed errors between the theoretical values and Adams simulation values are 0.29% to 3.29%, both lower than the 10% design threshold. This confirms the validity and reliability of the simulation results. The speed limit range covers the elevator speed requirement of 1.0–6.0 m/s, exhibiting excellent engineering applicability.
- (4) Finally, it should be pointed out that ANSYS/ADAMS commercial simulation software is difficult to be able to accurately complete complex friction modeling. Therefore, the simulation model in this paper ignores friction to improve computational efficiency and highlight core dynamic characteristics. This idealized assumption is reasonable under standard operating conditions. However, in long-term service, the problems of increased friction and component wear caused by lubrication failure have not been covered by the model, and quantitative modeling is a difficult point for subsequent research. To ensure the triggering accuracy of the speed limiter and the reliability of elevator operation, regular maintenance is required.

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References

- Wang, C.; Feng, S.C. Research on Collection and Preprocessing of Multisource Heterogeneous Elevator Data. In Proceedings of the 2020 IEEE International Conference on Power, Intelligent Computing and Systems (ICPICS), Shenyang, China, 28–30 July 2020; pp. 490–493.
- Gao, J.; Teng, Y.F.; Yuan, W.C. Applied Technology in the Test of Elevator Running Mileage (dynamic) of Scientific Significance. *Adv. Mater. Res.* **2014**, *1022*, 277–281.
- Chen, X.; Li, X.D. Android-based non-contact elevator overspeed governor speed measuring system. In *Proceedings of the 2016 3rd International Conference on Mechatronics and Information Technology*; Atlantis Press: Dordrecht, The Netherlands, 2016; pp. 67–71.
- Yang, G.X.; Liang, H. Condition Monitoring and Fault Diagnosis for an Antifalling Safety Device. *Shock Vib.* **2015**, *2015*, 286781. [[CrossRef](#)]
- Zhu, F.; Wang, Y.; Yang, K.; Wang, S.; Zhang, Q. Dynamic analysis and experimental study of direct-acting centrifugal speed limiter. *Proc. Inst. Mech. Eng.* **2025**, *239*, 1177–1189.
- Lonkwa, P.; Szydło, K. Selected parameters of the work of speed limiter in line straining system in a frictional lift. *Adv. Sci. Technol. Res. J.* **2014**, *8*, 73–77.
- Gradziel, M.J.; Holgersson, K.J. Mechanisms for Lowering Tethered Payloads: Lessons Learned from the Mars Exploration Program. In Proceedings of the 2008 IEEE Aerospace Conference, Big Sky, MT, USA, 1–8 March 2008; pp. 1–20.
- Tian, R.; Ma, J.X.; Ge, Y. Design of the Integrated Device New Speed Governor and Safety Gear. In *Proceedings of the Advanced Manufacturing and Automation X 10*; Springer: Singapore, 2021; pp. 364–370.
- Cenitagoya, A.; Ramirez, A.; Cabello, M. Novel Elevator Overspeed Governor Design and Multibody Dynamic Analysis. *J. Test. Eval.* **2022**, *50*, 2410–2421. [[CrossRef](#)]
- Xie, Z.; Liu, G.Y.; Wang, Y. Design and Analysis of a Novel Mini-type Centrifugal Friction Speed Limiter. *Appl. Mech. Mater.* **2014**, *496*, 677–682.
- Kutsepoleiko, A.V. Geometrical Analysis of the Hinge Mechanism of a Centrifugal Lift Speed Limiter With Two Output Links. *Mech. Mach. Mech. Mater.* **2024**, *4*, 61–69. [[CrossRef](#)]
- Zhang, M.; Laliberté, T.; Gosselin, C. Design and Static Analysis of Elastic Force and Torque Limiting Devices for Safe Physical Human-Robot Interaction. *J. Mech. Robot.* **2017**, *9*, 021003. [[CrossRef](#)]
- Ahmad, S.S.; Tom, L.; La Rocca, A.; La Rocca, S.; Vakil, G.; Gerada, C.; Benarous, M. Torque Limiters for Aerospace Actuator Application. *Energies* **2022**, *15*, 1467. [[CrossRef](#)]
- Micheau, P.; Oddo, R.; Lecours, G. Engine speed limiter for watercrafts. *IEEE Trans. Control Syst. Technol.* **2006**, *14*, 579–585. [[CrossRef](#)]
- Song, Y.P.; Chen, C. Design of Structural Parameters of Centrifugal Elevator Overspeed Governors. *Indones. J. Electr. Eng.* **2014**, *12*, 2480–2488. [[CrossRef](#)]
- Feng, S.C.; Chen, J.; Zhang, Y.B. Discussion on Improving Safety in Elevator Management. In Proceedings of the 2020 2nd International Conference on Machine Learning, Big Data and Business Intelligence (MLBDBI), Chengdu, China, 23–25 October 2020; pp. 195–198.
- Lin, H.P. Case analysis on trigger failure of elevator speed limiter. *Mech. Electr. Inf.* **2024**, 69–72.
- Shen, Y.; Sun, D.D. Case analysis of several elevator speed limiter failures. *China Elev.* **2021**, *32*, 62–64.
- Vermalle, J.C. *Proceedings of the 14th Aerospace Mechanisms Symposium: Proceedings of a Symposium Held at NASA*; Langley Research Center: Hampton, VA, USA, 1980; Volume 2127, p. 93.
- GB/T 7588-2020; Safety Regulations for Elevator Manufacturing and Installation. China Standards Press: Beijing, China, 2020.
- Liu, X.; He, Z. The Example Analysis of Super High Speed Elevator Car Noise Reduction and Vibration Reduction Design. *J. Langfang Teach. Coll. (Nat. Sci. Ed.)* **2013**, *13*, 38–41.

22. Chen, M.; Hong, Z.; Song, J.; Li, T.; Song, X.; Feng, Y. Vibration reduction optimisation design of the high-speed elevator car system based on multi-factor horizontal coupling vibration model. *IET Collab. Intell. Manuf.* **2024**, *6*, e70002. [[CrossRef](#)]
23. Xu, J.; Song, L.W.; Yang, Z. Mechanical vibration affects Gatling weapon shooting accuracy. *J. Phys. Conf. Ser.* **2021**, *1983*, 012004. [[CrossRef](#)]
24. Abdi, H. Partial least squares regression and projection on latent structure regression (PLS Regression). *Wiley Interdiscip. Rev. Comput. Stat.* **2010**, *2*, 97–106. [[CrossRef](#)]
25. Liu, R.X.; Kuang, J.; Gong, Q. Principal component regression analysis with SPSS. *Comput. Methods Programs Biomed.* **2003**, *71*, 141–147. [[CrossRef](#)] [[PubMed](#)]

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